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INTERNATIONAL APPLICATION PUBLISHED UNDER THE PATENT COOPERATION TREATY (PCT)

(51) International Patent Classification 5 : F01C 3/02

A1

(11) International Publication Number: WO 93/14299

(43) International Publication Date: 22 July 1993 (22.07.93)

(21) International Application Number: PCT/CA93/00005

(22) International Filing Date: 12 January 1993 (12.01.93)

(30) Priority data:
2,059,757 21 January 1992 (21.01.92) CA

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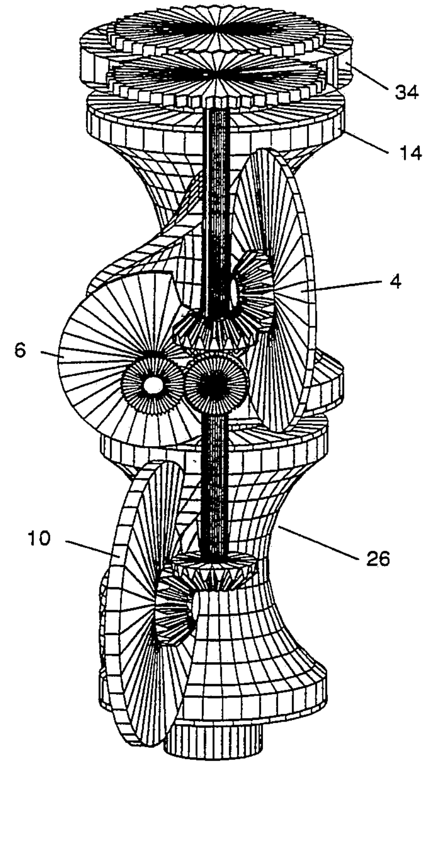
(81) Designated States: AT, AU, BB, BG, BR, CA, CH, CZ, DE, DK, ES, FI, GB, HU, JP, KP, KR, LK, LU, MG, MN, MW, NL, NO, NZ, PL, PT, RO, RU, SD, SE, SK, UA, US, European patent (AT, BE, CH, DE, DK, ES, FR, GB, GR, IE, IT, LU, MC, NL, PT, SE), OAPI patent (BF, BJ, CF, CG, CI, CM, GA, GN, ML, MR, SN, TD, TG).

**Published**
*With international search report.*
*With amended claims and statement.*

(54) Title: ROTARY ENGINE

(57) Abstract

An internal combustion engine composed of separate compression and combustion/expansion chambers (8) linked by a transfer duct (5), is disclosed. The compressor and expansion chambers include similarly shaped rotors (14, 26) and abutments (4, 10). Each of the rotors (14, 26) includes a vane (3, 9), and the rotors (14, 26) and abutments (4, 10) are rotationally coupled such that the vane (3, 9) enters into operative engagement with the associated abutment (4, 10) once during each revolution of the rotor (14, 26). Each abutment (4, 10) consists of a rotating disk having a gap (4g, 10g) which permits passage of the vane (3, 9) through the abutment (4, 10).



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# ROTARY ENGINE

## TECHNICAL FIELD

The present invention relates to internal combustion engines, and in particular to a rotary internal combustion engine.

## BACKGROUND ART

It is common in internal combustion engines to use parts that have reciprocating and/or eccentric oscillating movements. For example, the pistons and valves of a conventional auto or diesel engines have a generally linear reciprocating motion, while the rotor of Wankel-type engine has an eccentric oscillatory motion. In such engines, reciprocating or oscillating parts are subject to high accelerations, and thus experience large inertial forces, especially at higher revolution speeds. Typically, large counterweights are required to control vibrations caused by unbalanced inertial forces acting on the various moving components. The counterweights and rugged construction required for these reciprocating or oscillating parts considerably increase the weight of the engine. Furthermore, in spite of the use of balancing counterweights, these engines often suffer from vibration problems, which result in losses of efficiency.

Engines with reciprocating pistons typically require complex mechanisms with a large number of components, thus creating increased friction, wear, weight, production cost and maintenance.

Four cycle engines (i.e. conventional reciprocating piston engines) tend to be mechanically inefficient (i.e. have a low power to weight ratio) because each piston generates power over less than 1/2 revolution for every 2 revolutions of the crank shaft. Furthermore, low torque is developed at the beginning and end of the combustion cycle, due to the minimal moment-arm about the crankshaft at these points in the cycle.

In two cycle piston engines, the power to weight ratio is better than in four cycle engines, but these engines often suffer from inadequate exhausting of burned gasses and the resultant mixing of exhaust gasses with incoming fuel/air mixture.

In reciprocating piston engines, the use of intake and exhaust valves are required to control the flow of gasses through each cylinder. For example, the exhaust valve serves to constrain the release of combustion gasses, with the result that exhaust gasses must be exhausted from the cylinder within 1/2 revolution of the crankshaft.

In single stage engines (where gasses are compressed and burned in same chamber), external super-chargers are often employed to increase compression of the fuel/air mixture and thereby increase the power of engine. However superchargers consume engine power (thus reducing efficiency) and increase the complexity of the engine.

In reciprocating piston engines and most rotary engines, cooling of the piston (or rotor, as the case may be) is difficult because of shape and/or movement of the piston. In these engines, cooling is generally limited to the walls of the combustion chambers, thus leaving the engine vulnerable to thermal-expansion problems and overheating of internal parts.

In order for a conventional internal combustion piston or rotary engine to operate, highly effective seals must be provided. These seals are critical, particularly in the combustion chamber, where gas pressures prior to combustion can be in the range of 10 to 15 atmospheres, and where the gas pressure after combustion can be as much as a factor of ten higher. However, efficient operation of a conventional piston or rotary engine requires that the engine match (as closely as possible) the thermodynamic "Carnot-cycle", which demands that combustion occur within a constant volume. It will be apparent that leakage from the combustion chamber, particularly during and immediately following combustion, effectively prevents constant-volume combustion,

and thus seriously affects the engine efficiency. However, a problem common to most rotary piston engines is the construction of effective sealing components. The seals usually wear out rapidly, and in some cases are very difficult to build (for example, where inter-engaging rotors or vanes are employed).

## SUMMARY OF THE INVENTION

An object of the present invention is to provide an internal combustion engine which avoids many of the above-noted deficiencies.

According to an aspect of the invention, there is provided an internal combustion engine comprising: at least two rotary units, each of said units comprising a profiled rotor rotatably mounted on a shaft within a cylindrical housing, the profile of said rotor being adapted to operatively receive therein a sector of a rotatable disc-shaped abutment having an axis of rotation substantially perpendicular to said rotor, which abutment includes a substantially rectangular cutout and is rotationally coupled to said rotor so that the rotor and the abutment rotate at the same rate, said rotor further including a vane having a generally twisted-helical shape whereby the vane engages into the cutout of the abutment during each revolution of said rotor; and a transfer duct for conducting compressed gasses between respective ones of said rotary units.

According to a preferred embodiment of the present invention, two rotary units are coupled together by a shaft, and connected by a transfer duct. In this embodiment, a first rotary unit operates as a compressor, while the second unit operates as a combustion chamber. In addition, a rotary valve may be used to control the passage of air through the transfer duct.

In the following description, the rotary unit which operates as a compressor, will be referred to as the compressor or compression stage, while the rotary unit which

operates as a combustion and/or expansion chamber, will be referred to as the expansion chamber.

According to the present invention, all of the primary moving parts are rotary, thus effectively eliminating component stress and vibration due to non-circular movements. As a result, the weight of the engine can be reduced because the reduction of inertial stresses, due to the elimination of reciprocating and/or eccentric motions, permits lighter construction of the supporting structures of the engine, and limited use of counterweights.

The simplicity of a true rotary engine reduces mechanical complexity and the number of moving parts thus reducing friction, wear, weight, cost of production and maintenance.

The engine according to the present invention is capable of generating power over more than 2/3 of each revolution of the shaft. Additionally, maximum torque is obtained throughout the combustion/expansion phase because the moment arm of the expansion thrust (against the face of the expansion chamber vane) remains constant throughout the expansion cycle. In effect, the advantages of the two cycle and four cycle engines are combined in the present invention, as exhaust gasses are not mixed with intake gasses within the combustion chamber, and the engine generates power during each revolution of the shaft.

Within the engine of the present invention, fresh air is drawn into the compressor stage; compressed to elevate its pressure and temperature; and then transferred to an expansion phase. During the transfer and expansion processes, fuel is injected into the air flow, ignited, and the burning air/fuel mixture allowed to expand at substantially constant pressure. In this respect, the engine of the present invention operates approximately according to the "Joule Cycle", which is well known to be the thermodynamic cycle by which conventional gas-turbine engines operate. The present invention retains much of the simplicity and mechanical efficiency of the gas-turbine engine (in terms of the small

number of moving parts) but has significant advantages over gas turbine technology. In particular, the engine according to the present invention is capable of maintaining substantially higher compression ratios than is possible with conventional gas turbines, thus the operating efficiency is similarly much higher. Additionally, the engine of the present invention is capable of operation with substantially less air flow, and at substantially lower operating speeds, than is required for successful operation of a gas turbine.

The engine according to the present invention does not require an exhaust valve, because the interaction between the vane and abutment of the expansion chamber provides a physical separation between the gasses currently expanding and/or burning within the expansion stage, and exhaust gasses produced during the previous rotation cycle of the engine. Thus the exhaust gasses can be expelled continuously, without being constrained by the use of an exhaust valve. The elimination of exhaust valves reduces the number of moving parts.

The use of separate compression and expansion stages permits the use of a compressor stage having (for example) a larger volume than the combustion stage, thus facilitating increased compression ratios in the combustion/expansion stage.

The rotors, vanes, abutments and housings can be continuously cooled and maintained at a similar temperature, thus preventing the overheating of internal parts. Maintaining all parts at similar temperatures reduces stress and adjustment problems caused by variations in the thermal expansion of adjoining parts.

BRIEF DESCRIPTION OF THE DRAWINGS

The present invention will now be described, by way of example, with reference to the appended drawings, in which:

Figure 1 shows an embodiment of a rotor and an abutment according to the invention;

Figure 2 shows an embodiment of an engine according to the present invention, illustrated in a first position in its operating cycle.

Figure 3 shows the embodiment of Figure 2 at a second position in its operating cycle;

Figure 4 shows the embodiment of figure 2 at a third position in its operating cycle;

Figure 5 shows the embodiment of Figure 2 at a fourth position in its operating cycle;

Figure 6 illustrates the exterior features of the housing of the embodiment of Figure 2;

Figure 7 presents an illustration of the curve mapped out by the rotation of an abutment in association with a rotating rotor;

Figure 8 illustrates the alignment between the face of an abutment and the centre axis of a rotary unit employed as a compressor;

Figure 9 presents a cross-sectional view taken through a rotary unit according to the invention;

Figure 10 illustrates an abutment housed within an abutment housing;

Figure 11 presents a partial cross-section view of a vane and a rotary unit housing, illustrating sealing and lubrication arrangements;

Figure 12 presents an illustration of an embodiment of a transfer duct according to the present invention;

Figure 13 presents a side view of the transfer duct illustrated in Figure 12;

Figure 14 presents a diagrammatic cross-sectional view through a rotary unit according to the present invention, employed as an expansion and combustion chamber;

Figure 15 presents an illustration of the rotor and vane of the rotary unit of Figure 14;

Figure 16 illustrates an abutment and an abutment housing employed in the rotary unit illustrated in Figure 14;

Figures 17A - 17D illustrate various embodiments of mechanical seals employed to provide sealing between a rotor, abutment and vane in an embodiment of the present invention;

Figures 18A and 18B illustrate the relationship between an abutment edge and adjacent rotor and vane surfaces in an alternate embodiment of the present invention.

Figure 19 illustrates an embodiment of shaft and gear arrangements used to provide synchronised rotation of the various components in an embodiment of the present invention.

Figure 20 presents an illustration of the effect of increasing the size of a rotor; and

Figure 21 presents an illustration of the effect of increasing the size of an abutment.

## DETAILED DESCRIPTION OF THE PREFERRED EMBODIMENTS

It should be noted that the following discussion of example embodiments of the invention includes discussions of possible mechanical sealing, lubrication and cooling arrangements which, in a practical engine, would be required to ensure smooth and reliable operation. However, it will be understood that the precise design details of these arrangements are considered to be in the domain of those skilled in the art. Accordingly, it will be understood that the possible mechanical sealing, lubrication and cooling arrangements, described in conjunction with the example embodiments, are not to be taken as being limitative of the present invention.

According to an embodiment of the present invention, the compression and expansion rotors are coupled together and therefore rotate simultaneously (see Figures 2 to 5). However, to simplify the description of the compression, transfer, expansion and exhaust phases of operation, each phase is explained separately.

Figure 1 shows a rotary unit and an associated abutment. In this case, the rotary unit operates as a compressor, and the rotor is illustrated at an intermediate (relatively early) point during the intake/compression cycle.

As the compression rotor 14 rotates (in the direction of the arrow), fresh air is drawn into the expanding intake volume 2 of the compression stage, through an intake port 1. At the same time, air which entered the compression stage during the previous rotation of the compression rotor 14 is compressed between the advancing leading face 47 of the compressor vane 3 and the high-pressure side 4h (see Figure 2) of the compressor abutment 4.

Figure 2 shows a compressor, transfer duct and expansion chamber at a point near the end of a compression cycle. A rotary transfer valve 6, which is rotationally synchronized with the compressor rotor 14, seals the transfer duct inlet 7 between the compression stage and the transfer duct 5.

Referring to Figure 2, at the end of the compression cycle the compressor rotor 14 and abutment 4, which are also rotationally synchronized, are in a position where the compressor vane 3 is about to engage into a substantially rectangular compression abutment gap 4g. The rotary valve 6 is also in a position where the valve 6 is about to open the transfer duct inlet 7.

At this point, the combustion vane 9 almost completely blocks the passage between the transfer duct 5 and the expansion chamber 8, which is (in this position) confined to a very small volume defined by the expansion vane 9 and the high pressure side 10h of the expansion abutment 10, which engaged with each other at a slightly earlier point in the rotation of the engine.

As illustrated in figures 2 to 5, the compression stage can have a larger volume than the expansion chamber, thereby allowing for further compression of the gasses during transfer, and to compensate for the volume of the transfer duct 5.

The compression ratio for the engine can be calculated according to:

$$C = \frac{V1 - (V2 + V3)}{V4 + V5 + V6}$$

Where: C = Compression ratio
V1 = Volume of compression chamber
V2 = Volume of vane
V3 = Volume loss due to intake port
V4 = Volume remaining in compressor at the end of transfer cycle (as explained below).
V5 = Volume of transfer duct
V6 = Volume of expansion chamber at end of transfer cycle (explained below)

The volumes V1 - V6 can be calculated from geometrical considerations. By using the above relationship, and assuming that compression loss due to leakage is negligible, the compression ratio of the example engine illustrated in Figures 2 through 6 can be determined to be approximately 7.5:1. It will be understood that under true operating conditions, a certain degree of "blow-by" (or gas leakage around the abutments and vanes) is inevitable. In an embodiment of the present invention, the amount of blow-by is restricted by the use of sealing elements adapted to provide a mechanical seal between moving parts.

In an alternate embodiment, the blow-by is limited by the shape of the edges of the abutments and vanes and by running the engine at sufficient speeds to limit the time available for air leakage so as to reduce the compression losses. In this alternate embodiment, a narrow running clearance is maintained between the various components. Thus while the amount of blow-by is substantially higher than in a mechanically sealed embodiment, the internal friction and wear of the engine is greatly reduced. Additionally, the mechanical simplicity of the engine is enhanced by the elimination of mechanical sealing elements. Maintenance of an operating clearance between components also has the advantage of obviating the problems associated with lubricating vane and abutment surfaces, because frictional contact between these surfaces is eliminated.

Figure 3 illustrates the engine illustrated in Figure 2 at an intermediate point during the transfer phase of operation. At this point, the rotary valve 6 is open, thus

allowing compressed air (or fuel-air mixture) to pass through the transfer duct 5 and into the enlarging expansion chamber 8 defined by the expansion abutment 10 and expansion vane 9.

Note that Figure 3 also shows a different arrangement of the structural connection 38 between the compression and expansion stage housings.

Figure 4 illustrates the engine illustrated in Figure 2 at the end of the transfer phase of operation. At this point, rotation of the rotary valve 6 has closed the transfer duct inlet 7. The compressed air (or fuel-air mixture) fills the transfer duct 5 and the continuously enlarging expansion chamber 8 defined between the expansion vane 9 and the expansion abutment 10.

Fuel can be mixed with the air either prior to entering the compressor, or injected in the transfer duct 5 during the transfer cycle. A fuel/air mixture can enter in the compressor via a conventional carburettor, which facilitates improved fuel vaporization and mixing of gasses before entering the transfer duct and expansion chamber. In this case it is also advantageous to use either fuel which in a gaseous form at operating temperatures and pressures (for example, propane, natural gas etc.), to avoid dissolving of lubricant used in compression chamber parts. Alternatively, two cycle oil can be added to the fuel to serve as lubricant, or the liquid fuel could be heated to promote vaporisation. Otherwise a gasoline resistant lubricant should be used to lubricate compression stage components.

Alternately, fuel (liquid or gaseous) can be injected into the transfer duct 5 during the transfer phase. In this case, the injector should be placed as close as possible to the transfer duct inlet 7 to allow more time for mixing of the fuel and air, prior to entering the expansion stage.

At the end of the transfer cycle, a spark plug 11, located in the transfer duct 5, ignites the fuel-air mixture so as to initiate expansion. A second spark plug 59 (see Figure 6) can be installed within the expansion chamber wall

(a suitable recess can be provided to avoid obstructing the passage of the expansion vane), close to the expansion abutment 10, to facilitate direct ignition of the fuel within the expansion chamber.

Ignited gasses, which are sealed within the expansion chamber 8 by the expansion abutment 10, the rotary valve 6 and the expansion vane 9, exert a high and substantially constant, pressure on the trailing face 48 of the expansion vane 9, while the leading face 49 of the expansion vane 9 is subject to much lower (approximately atmospheric) pressure. The pressure difference across the expansion vane 9 produces a torque about the expansion rotor 26 and output shaft 13, thereby forcing the expansion rotor 26 to rotate in the direction indicated by the arrow in Figure 4.

Thrust generated on the expansion vane 9 decays towards the mid-point of the next transfer cycle (see Figure 3) as the gasses in the expansion chamber expand to near atmospheric pressure, and both leading 49 and trailing 48 faces of the expansion vane 9 are subject to approximately the same pressure. Thus thrust is generated on the expansion vane 9 for approximately 235° with a decrease in thrust for the following 60°, as the expansion vane 9 progressively engages with the expansion abutment 10. Power is therefore generated over more than 2/3 of each revolution.

An exhaust port 12, is located in the lower part of the expansion chamber, close to the expansion abutment 10, as shown in Figures 5 and 6. The exhaust port 12 does not have a valve, and thus burned gasses are expelled continuously through the exhaust port 12 as the leading face of the expansion vane 9 advances towards the port 12.

Figure 6 illustrates the exterior of the compressor and expansion chamber housings, clearly indicating the relative positions of the compressor inlet 1, transfer duct inlet 7 and outlet 25, and the exhaust port 12. In addition, the slots 44 and 45, through which the compressor and expansion abutments respectively are mounted, are illustrated.

Furthermore, the position of the rotary valve housing 43, with respect to the transfer duct inlet 7 and the compressor abutment is illustrated. Finally, an opening 59, can be provided to facilitate installation of a spark plug so as to permit ignition of fuel/air mixture within the expansion chamber, as discussed previously.

The offset between the compression and expansion components (i.e. the angle between the compression and expansion abutments) is dictated by the length of the transfer cycle and the synchronization of respective compression and expansion rotors and vanes. In the embodiment shown in Figures 2 through 6, the transfer cycle begins before the expansion vane has completely crossed the outlet 25 of the transfer duct 5 (to enclose the expansion chamber), so as to permit the incoming compressed gasses to force some of the exhaust gasses within the transfer duct to be vented past the expansion vane, and thus reducing the amount of exhaust gasses being introduced into the expansion chamber 8 from the transfer duct 5.

The accurate alignment of the abutments with the respective rotors is of prime importance, because the alignment determines the degree to which an adequate seal can be maintained to prevent excessive blow-by of gasses around the abutments, with the associated losses in compression and efficiency and utilisation of expansion chamber pressure.

Figure 7 illustrates the curvature of the faces of the compression and expansion vanes. The rotation of the rotor and abutment are synchronised, so that the contour of the vane must correspond with the motion of an edge of the abutment gap as the rotor and abutment rotate together. As illustrated in Figure 7, a point (a) at a given radius from the centre of the abutment (a point at the perimeter of the abutment is illustrated) maps out a curve F as the rotor and abutment rotate in the directions indicated by respective arrows. Locations b, c, d and e on the curve F indicate various positions of the point (a) at various stages during the rotation, and illustrate the relationship between the

plane of the abutment face G and the vane face at those locations. In particular, it can be seen that the angle between the abutment face G and the vane varies as the vane passes through the abutment.

Because the angle between the abutment face G and vane face F varies along the path followed by the two components, it is not possible for both leading and trailing faces of a vane to conform with the leading or trailing faces of an abutment.

The alignment between the abutments and rotors in the engine according to the invention is such that a seal must be maintained between one edge on the high-pressure side of an abutment and one face of the corresponding vane. In particular, the compressor abutment 4 must maintain a seal against the leading face of the compressor vane 3, while the expansion abutment 10 must maintain a seal against the trailing face of the expansion vane 9. Referring to Figure 8, the high pressure side 4h of the compression abutment 4, which faces the leading face of the compression vane 3, is aligned with the centre axis of the compression rotor 14. Conversely, the side of the expansion abutment which faces the trailing face of the expansion vane 9, is aligned with the centre axis of the expansion rotor.

Figure 9 shows a cross-section of the compression rotor 14, the compression stage wall 15 and rotor sealing rings 16. The rotor can be formed as a hollow shell, contoured 17 to correspond to the shape of the abutment. Counter-weights 18 can be placed inside the rotor, opposite to the compressor vane 4, to balance the weight of the vane, thus eliminating unbalanced forces, and associated vibrations. The rotor rings-seals 16 are stationary (i.e. do not rotate with the rotor), a portion of each ring seal 16 being engaged into the chamber housing 15. The rotor seals 16 can be made of steel to allow for flexibility and wear resistance properties. The ring-seal gap 53 is located in line with the abutment 4 to prevent any leakage of gasses and permitting

lubrication of both rotor seal and abutment seal at the same location.

Referring to Figure 10, the gap 4g in the compressor abutment 4 is shaped to accept the compressor vane 3 while passing through the abutment 4, with only one edge 50 (on the high-pressure side 4h) closely fitting the face of the vane. Since the curves along inside and outside edges of the vane 3 are different, the inside faces 21 of the abutment gap 4g can be twisted in order to more closely conform to the face of the vane. However, because only one edge 50 must maintain a seal with a corresponding face of the vane 3, the precise geometry of the remaining surfaces of the abutment gap is not critical, beyond the fact that they must not interfere with passage of the vane 3.

A seal, made of compressible resistant material, is disposed near the sealing edge 50 of the gap 4g and along the circumferential edge 22 of the abutment 4, in order to seal the gaps between the abutment, rotor and vane. The seals need to be compressible to provide effective sealing using a single seal between the abutment and the vane, as only the high-pressure face 4h of the abutment 4 is aligned with the centre axis of the rotor, and only one edge 50 of the abutment 4 exactly follows the curvature of the face of the vane 3. Using a compressible material permits a wider area of seal contact, and allows for compensation of thermal expansion of parts.

A preferred material for abutment seals would be a silicone-teflon base substance for it's heat resistance and lubrication properties. The material should however be hard enough to resist pressures exerted by the compressed gasses and needs to be only slightly compressible as the gaps to be filled due to thermal expansion are extremely small (hundredths of millimetre).

A housing 23 is placed around the abutment to keep pressure on the seal while not in contact with the rotor to prevent excessive deformation of the flexible seal at high revolution speeds. The faces of the abutments are lubricated

by using oil-grooves inside the housing walls. The gap faces are lubricated by injecting oil (or lubricant) through small holes located on the inside face of the abutment gap 57 as the gap enters the housing. Centrifugal force will cause the oil to flow along the entire face of the seal.

The abutment housing 23 is attached to the wall of the compression chamber and the abutment 4 enters the chamber through a slot 44 (see figure 6) in the chamber wall. A seal 56 (see Figure 8) is placed between the abutment housing and the compression face 4h of the abutment to prevent leakage through the abutment gap while it is inside the housing.

The abutment can be balanced by creating hollow areas inside the abutment opposite the gap, or by using external counterweights attached to the abutment shaft. These abutment counterweights would have to be located outside the abutment housing as the housing needs to closely fit the abutment to permit proper heat transfer for cooling purposes.

The vane 3 should be as thin as possible while remaining strong enough to sustain pressures exerted by compressed gasses and resist centrifugal forces caused by the high revolution speeds of the rotor.

A wear resistant rigid seal 19 ( for example a metal alloy) is placed between the vane and the compressor wall 15 (see Figure 11) and the expansion gap is filled with compressible material 20 (similar to the abutment seals) to prevent escape of compressed gasses past the face of the abutment during compression.

An oil groove 52, filled with porous rigid material is located in the wall of the compression chamber 15 to oil the vane seal 19 as it passes over the groove 52. The oil groove is located close to the compressor inlet port 1 to avoid being submitted to high pressures which would impair the lubrication of the vane seal 19.

The rotor, vane, abutment, chamber wall and housing can be made of metal having similar thermal expansion coefficients to avoid problems related to differences in thermal expansion. Iron and steel alloys are preferred for

this embodiment because of their low thermal expansion coefficient, good resistance and rigidity, ease of casting (important mainly for the rotor and vane) and low production cost. On the other hand, aluminium alloys (such as silicon and copper) are better heat conductors, are lighter and can attain resistance approaching that of steel but have higher thermal expansion coefficients and are more expensive.

Referring now to Figure 12, the rotary valve 6 is constructed as large as possible to permit the valve gap 24 to be in the full open position as long as possible in order to permit maximum free flow of gasses between the compressor and expansion chamber through the transfer duct inlet 7.

A valve seal between the valve 6 and the compressor, is not required as the valve rests tightly against the compression chamber face of the valve housing 43 (see Figure 6) and the valve overlaps the transfer duct inlet 7. The valve 6 is therefore forced to bear against the compressor side of the housing 43 by the pressure exerted by the expansion gasses, except (possibly) at the end of the compression cycle where pressure on both sides could become similar. As a security measure, a spring loaded washer could be placed in the valve housing on the transfer duct side to force the valve 6 to continuously bear against the compressor side of the housing 43.

A spark plug 11 is placed in the transfer duct 5 as close as possible to the exit port 25 in order to ignite the fuel-air mixture efficiently. Figure 13 provides a side view of the transfer duct.

The rotary valve 6 can be balanced by using methods similar to that described in connection with the compression abutment. Lubrication of the valve 6 can be accomplished by providing oil grooves on the valve housing. A seal is also placed inside the valve housing, around the entrance port 7, on the side adjacent to the transfer duct, to prevent expansion gasses from exerting a back pressure on lubricating oil. This seal has to be larger than the valve gap so the

seal is always resting on a portion the valve large enough to avoid any interference between the valve and the seal.

In an alternative embodiment, the rotary valve 6 can be replaced by an oscillating valve (such as a "poppet" valve) which alternately opens and closes the passage between the compression chamber and transfer duct. An oscillatory valve member of this type, which remains stationary during the combustion phase, reduces internal friction and simplifies construction of sealing components which are associated with a rotating valve.

In addition, a further valve (either rotary or oscillating) can also be mounted to open an close the passage between the transfer duct and the expansion chamber. Such a valve permits the advance of the transfer cycle, while avoiding leakage of fresh fuel/air mixture into the expansion chamber, thereby permitting ignition to occur earlier (such as is illustrated in Figure 5), while the fuel/air mixture is in a higher state of compression and not expanding as rapidly as would be the case in the embodiment illustrated in Figure 4.

Referring to Figure 14 and 15, the shape of the expansion rotor 26, is similar to that of the compression rotor, but the expansion rotor is smaller and is of heavier construction to resist internal expansion chamber temperature and pressure. Sealing rings can be placed at each end of the rotor 27, the arrangement and lubrication thereof being similar to the compressor rotor seals described previously. Similarly, counterweights can be located within the expansion rotor in a manner similar to that described above, to balance the expansion vane. However, care must be taken to ensure that the counterweights do not interfere with cooling of the rotor.

The expansion vane 9 is shaped as described above in connection with the compression vane 3, but is generally thicker, and is provided with internal cooling channels to permit effective cooling of the expansion vane.

Referring to Figure 15, a seal 28 is fitted in a groove along the peripheral edge of the vane 9 between the vane 9 and the expansion chamber wall, to compensate for possible difference in expansion between the vane 9 and chamber wall, while maintaining an effective seal.

In an alternative embodiment, the vane 9 can be constructed as a separate component and mounted into a groove in the rotor 2. A tight fit between the vane and rotor groove substantially reduces blowby and play of the vane, while permitting adjustment of the rotor and vane due to thermal expansion. Sealing material within the groove can be used to further reduce blowby of compressed gasses. The vane 9 can also be extended and widened at each end to provide a sliding face in contact with the cylinder wall which can readily be lubricated as it is located outside the combustion (or compression) chamber. The vane portion within the combustion (or compression) chamber is slightly recessed from the sliding face (by approximately 0.001 inches or less) to avoid contact with the cylinder wall while providing and adequate seal, thus avoiding lubrication problems inside the chamber. A groove between the vane and vane extension (at both ends of the vane) can be used to prevent interference between the vane and sealing rings.

Referring to Figure 16, the expansion abutment 10 has the same general shape as the compression abutment 4, but has wider gap 10g to accommodate the wider expansion vane. The high pressure face 10h is aligned with the centre axis of the rotor, and a sealing edge 51 closely follows the trailing face of the expansion vane 9 in order to maintain an efficient seal in the expansion chamber 8.

The seals on sealing edge 51 of the abutment gap 10g, and the seals around the perimeter of the abutment 10 are preferably made of compressible heat and wear resistant material similar to that described for the compression components. The seals could also be made of hard and flexibly mounted (i.e. spring loaded) material. Figures 17(a) and 17(b) illustrate the use of compressible seals in the

compressor and expansion chamber abutment gaps 4g and 10g respectively. Rigid seals, as illustrated in Figures 17(c) and 17(d), can also be used. In the latter case, the seal movements can be limited by spring-loaded chokers which facilitate continuous adjustment of the sealing element, and thus ensuring an effective seal between a vane and abutment.

A housing 33 is placed around the abutment to keep pressure on the seal while not in contact with the rotor to either prevent excessive deformation of the compressible seal at high revolution speeds or keep a tight fit on the spring loaded seal. A lining 46 can be used within the housing 33 (see Figure 14), to maintain alignment between the seal and the rotor. A seal is placed in the abutment housing 33, between the abutment housing 33 and the high pressure face 10h of the abutment 10 to prevent leakage through the abutment gap 10g while inside the housing. The abutment housing 33 is attached to the wall of the expansion chamber and the abutment 10 enters the chamber through a slot 45 (see Figure 6) in the expansion chamber wall.

The abutment 10 can be counterbalanced and all seals are lubricated in the same manner as described above with respect to the compression stage.

The expansion rotor, vane, abutment and cylinder wall, as well as the rotary valve and transfer chamber are preferably cooled to keep thermal expansion to a minimum thus avoiding distortion and seizing due to differences in thermal expansion rates between various adjoining parts. Figure 19 illustrates a centrifugal fan 34, directly coupled to the compressor rotor, which can be used to cause a circulation of air through and around the engine. The precise arrangements for cooling, either by air-cooling, a circulating cooling liquid (such as water, or at least predominantly water), or a combination of these, is within the domain of one skilled in the art, and is therefore not described in detail here.

The above described sealing, lubrication and cooling arrangements are generally related to a mechanically sealed embodiment of the invention. As previously mentioned, in an

alternate embodiment, mechanical seals between the rotor, vane, and abutment are not employed. In this embodiment, gas flow through the narrow gaps between these elements is restricted by the shape of the edge of the abutment and vane, and by maintaining a narrow clearance between adjacent surfaces. In particular, Figures 18A and 18B diagrammatically illustrates the relationship between the edge of the abutment and the rotor (or vane) surface and between the edge of the vane and the interior surface of the rotor housing. The geometry illustrated in Figures 18A and 18B utilises the hydrodynamic principle that a sharp-edged inlet to a duct increases the resistance to fluid flow through the duct. Thus the high pressure side of the abutment (and vane) is provided with a sharp, chisel-like, edge. The peripheral surface of the abutment (or vane) may be parallel to the adjacent surface, or may be angled slightly so that the clearance increases towards the low-pressure side of the abutment (or vane), as illustrated. In the latter case, the peripheral surface should be close to parallel in order to provide sufficient material to ensure adequate strength at the edge, and to facilitate adequate conduction of heat away from the edge. In addition, a narrow operating (or running) clearance is maintained to prevent frictional contact between the edge and the adjacent surface. This clearance should be as narrow as possible, while being large enough to ensure that a minimum clearance will be maintained in spite of thermal expansion.

In the alternate embodiment, a certain amount of compression loss is inevitable. However, the adverse effects of the compression loss can be at least partially compensated for by increasing the rotation speed of the engine, thereby reducing the amount of time available for compression loss during each rotation cycle of the engine. Additionally, as the engine heats up, thermal expansion of the rotors, vanes, abutments and housings reduces the width of the running clearance. Thus the amount of compression loss reduces as the engine is brought up to normal operating temperature.

Elimination of mechanical seals and maintenance of a working clearance, substantially simplifies the engine, and reduces friction, thereby obviating the need for lubrication of the edges of the abutments and vanes. However, the higher operating speeds, and the fact that the engine is now running "dry" (i.e. without lubrication) can be expected to increase the operating temperatures of the engine. In addition, because the amount of compression loss is directly related to the clearance between components, which is in turn dependent on thermal expansion rates, careful selection of heat-tolerant materials is very important. For example, the abutments, rotors and vanes can be fabricated from cast ceramics of the type currently being used in advanced gas-turbine engines, and (experimentally) in automotive piston engines. These materials exhibit low thermal expansion rates, high resistance to thermal shock, and have also been shown to maintain dimensional tolerances and structural integrity at extreme temperatures.

Finally, the superior thermal resistance of cast ceramics means that the abutments, and rotor vanes can be operated substantially without cooling. Thus in this alternative embodiment, cooling of the engine can essentially be limited to the circulation of air around the exterior of the engine (possibly using cooling fins to increase heat transfer rates), and (possibly) the interior of the rotors. Alternatively, liquid cooling can be used, in this case by circulating cooling liquid (such as, for example, a conventional coolant consisting at least predominantly of water) through a jacket surrounding the transfer duct and expansion chamber.

As shown in Figures 2 through 5, the compression 14 and expansion 26 rotors are directly linked together. The expansion 10 and compression 4 abutments and rotary valve 6 are linked by suitable gears and shafts to the rotors 14 and 26, so that all moving parts are synchronized. For example, the shaft and gear arrangements shown in Figure 19 illustrates an embodiment of the invention which includes a single

synchronising shaft, and appropriate gears to ensure that the compressor and expansion chamber abutments 4 and 10, as well as the rotary valve 6, all rotate together at the correct rotation speeds by being driven by a gear coupled to the output shaft 13 of the engine. The precise design of the gears, shafts, bearing arrangements, lubrication and gear housings etc. is considered to be well within the capabilities of those skilled in the art, and therefore is not discussed in further detail here.

The rotors are mounted within the housings using suitable bearings. Seals (for example ring seals) are provided at either end of the rotors to prevent escape of gasses around either end of the rotor. Here again, the precise design of the rotor bearings and end-seals, including supporting structures, housings, lubrication etc. is well within the capabilities of those skilled in the art, and therefore is not discussed in detail here.

It will be apparent from the forgoing that there will be a variety of ways in which the engine of the invention can be modified. The following is a brief description of some of the modifications which are possible.

As shown in Figure 20, the size of the compression chamber can be modified by increasing or decreasing the axial length of the rotor 14 without increasing the diameter of the abutment 4, and thereby without affecting the alignment of compression and expansion abutments and their associated gearing system.

The shape of expansion and compression chambers shown in Figures 2 to 5, represents a preferred embodiment of the invention. It is possible to use other shapes of vane and chamber or proportions between vane, rotor and abutment. However, variations in these components will influence the performance and characteristics of the engine as explained below.

Figure 21 shows the effect of increasing the diameter of the abutment from A to A1. Since the rotation of the abutment is synchronized with the rotor (and vane) B, the

length of the transfer cycle is proportional to the angle n,n1 formed by the contact between the rotor and the abutment. Thus increasing the diameter of the abutment decreases the circumferential length of the transfer cycle without increasing significantly the volume of the chamber. Figure 18, shows an increase of the abutment diameter of 20%, which produces a decrease in the circumferential length of the transfer cycle by 11% (note that n>n1) and an increase in the volume of the compression chamber by only 5%. A shorter transfer cycle would increase internal pressure in the compression chamber resulting in excessive pressures on the seals thus increasing the risks of loss of compressed gasses.

Conversely, increasing the size of the rotor (as shown in Figure 20), has the effect of increasing the circumferential length of the transfer cycle and increasing the volume of the chamber. Increasing the size of the compression rotor would increase the compression ratio, but an increase of the expansion rotor could lead to problems related to thermal expansion of the abutment inside the cylinder and increase the difficulty of sealing the gap between the abutment and rotor. It would also create problems related to a shorter cooling cycle of the abutment and would require a stronger abutment due to the increased area subjected to expansion pressure.

An increase in the diameter of both compression and expansion rotors would increase the volume of the chambers, the circumferential length of the transfer cycle and increases the torque of the engine. On the other hand an increase in rotor diameter increases centrifugal force on the vane seal, the length of rotor seal and seal area of abutment face on the vane. These potential problems could, however, be compensated for by doubling the rotor rings and counterbalancing the vane seal.

Reducing the rotor to the shape of a disk, as described in United States Patent Nos, 3,841,276, 3,502,053, 3,674,982, 3,012,551 and 3,205,874 has the disadvantage of creating a chamber volume for a given diameter of rotor and

makes the sealing of the rotor extremely difficult, particularly where the abutment enters the rotor housing.

The compression and expansion stage walls could also be convex, thus providing each stage with a barrel shaped exterior appearance. The advantage of using convex chamber walls is the increase chamber volume with little increase in engine size. The disadvantages are related to increased problems of thermal expansion of a large expansion vane inside the chamber; the complexity of vane seal fitting with the rotor seal; the necessity of building the stage housings in two sections facilitate insertion of the rotor; and increased problems in cooling the vane due to it's complex shape.

It is also possible to use two or more abutments arranged around a rotor. However, this would yield little advantage as the total engine displacement would be the same, and the engine would go through two or more "dead" phases (corresponding to the transfer cycle), instead of only one. In order to increase engine output while maintaining continuous power (i.e. minimising any dead phase), the preferred embodiment uses a second set of compression and expansion stages, connected in line with the first set, and offset by 180° about the common shaft.

CLAIMS:

1. An internal combustion engine comprising:

at least two rotary units, each of said units comprising a profiled rotor (14, 26) rotatably mounted on a shaft (13) within a generally cylindrical housing, the profile of said rotor (14, 26) being adapted to operatively receive therein a portion of a rotatable disc-shaped abutment (4, 10) having an axis of rotation substantially perpendicular to said rotor (14, 26), which abutment includes a substantially rectangular cutout (4g, 10g) and is rotationally coupled to said rotor (14, 26) so that the rotor (14, 26) and the abutment (4, 10) rotate at the same rate, said rotor (14, 26) further including a vane (3, 9) having a generally twisted-helical shape whereby the vane (3, 9) operatively engages into the cutout (4g, 10g) of the abutment (4, 10) during each revolution of said rotor (14, 26); and

a transfer duct (5) for conducting compressed gasses between respective ones of said rotary units (14, 26).

2. An internal combustion engine as claimed in claim 1, wherein one of said rotary units (14, 26) operates as a compressor for compressing a gas, and a second one of said units operates as a combustion and expansion chamber for converting energy released by combustion of a fuel in the gas into mechanical torque on a shaft (13), said transfer duct (5) serving to allow compressed gas being transferred from the compressor to the combustion and expansion chamber.

3. An internal combustion engine as claimed in claim 2, wherein the gas is a mixture of air and a fuel.

4. An internal combustion engine as claimed in claim 2, wherein the gas is air, and a fuel is added during the transfer of compressed gas from the compressor to the combustion and expansion chamber.

5. An internal combustion engine as claimed in claim 2, 3 or 4, further comprising a substantially disc-shaped rotary valve (6) rotatably mounted in operative relation to the opening (7) between the compressor and the transfer duct (5), said rotary valve (6) being adapted to control the flow of gas from the compressor and into the transfer duct (5).

6. An internal combustion engine as claimed in any one of claims 1-4, wherein leakage of gasses past each said abutment (4, 10) is restricted by mechanical sealing elements (22, 50, 51, 56) disposed about the peripheral edge and the substantially rectangular gap of the abutment, thereby substantially closing a gap between the abutment and the adjacent rotor and vane surfaces.

7. An internal combustion engine as claimed in any one of claims 1-4, wherein leakage of gasses past each said rotor vane is restricted by mechanical sealing elements (19, 28) disposed about the peripheral edge of the vane, thereby substantially closing a gap between the vane and an interior surface of a rotor housing.

8. An internal combustion engine as claimed in any one of claims 1-4, wherein leakage of gasses past each said abutment (4, 10) is restricted by the geometry of the edge of the abutment, and further by maintaining a narrow clearance between the abutment edge and adjacent rotor (14, 26) and vane (3, 9) surfaces.

9. An internal combustion engine as claimed in any one of claims 1-4, wherein leakage of gasses past said vane (3, 9) is restricted by the geometry of the edge of the vane (3, 9) and further by maintaining a narrow clearance between the edge of the vane (3, 9) and an interior surface of a rotor housing.

10. An internal combustion engine as claimed in any one of claims 1-4, wherein leakage of gasses past each said abutment (4, 10) is restricted by the geometry of the edge of the abutment (4, 10), and further by maintaining a narrow clearance between the abutment edge (22, 50, 51) and adjacent rotor (14, 26) and vane (3, 9) surfaces, and wherein said narrow clearance is in the range of approximately 0.02 to 0.08 inches in width when the engine is cold.

11. An internal combustion engine as claimed in any one of claims 1-4, wherein leakage of gasses past said vane (3, 9) is restricted by the geometry of the edge of the vane (3, 9), and further by maintaining a narrow clearance between the edge of the vane 3, 9) and an interior surface of a rotor housing, and wherein said narrow clearance is in the range of approximately 0.02 to 0.08 inches in width when the engine is cold.

12. An internal combustion engine as claimed in any one of claims 1-4, further comprising a system of one or more gears and shafts adapted to cause rotation of each said abutment in synchrony with respective ones of said rotors.

13. An internal combustion engine as claimed in claim 1, further comprising cooling means (34) for removing excessive heat from within said engine.

14. An internal combustion engine as claimed in claim 13, wherein said cooling means (34) comprises means for circulating a cooling fluid around at least a portion of the exterior of said engine.

15. An internal combustion engine as claimed in claim 13, wherein said cooling means (34) comprises means for circulating a cooling fluid around at least a portion of the interior of said engine.

16. An internal combustion engine as claimed in claim 13, 14 or 15, wherein said cooling fluid is air.

17. An internal combustion engine as claimed in claim 13, 14 or 15, wherein said cooling fluid is at least predominantly water.

18. An internal combustion engine as claimed in any one of claims 1-4, wherein combustion of a fuel within said engine in initiated by means of a spark-ignition system comprising a spark plug (11) operatively disposed within a portion of the transfer duct (5).

AMENDED CLAIMS

[received by the International Bureau on 13 May 1993 (13.05.93) ; original claims 1, 10 and 11 amended ; new claim 19 added; other claims unchanged (3 pages)]

1. An internal combustion engine comprising:

at least two rotary units, each of said units comprising a profiled rotor (14, 26) rotatably mounted on a shaft (13) within a generally cylindrical housing, the profile of said rotor (14, 26) being adapted to operatively receive therein a portion of a rotatable disc-shaped abutment (4, 10) having an axis of rotation substantially perpendicular to said rotor (14, 26), which abutment includes a substantially rectangular cutout (4g, 10g) and is rotationally coupled to said rotor (14, 26) so that the rotor (14, 26) and the abutment (4, 10) rotate at the same rate, said rotor (14, 26) further including a vane (3, 9) having a generally twisted-helical shape whereby the vane (3, 9) operatively engages into the cutout (4g, 10g) of the abutment (4, 10) during each revolution of said rotor (14, 26), said vane (3, 9) extending around said rotor (14, 26) by less than 180 degrees; and

a transfer duct (5) for conducting compressed gasses between respective ones of said rotary units (14, 26).

2. An internal combustion engine as claimed in claim 1, wherein one of said rotary units (14, 26) operates as a compressor for compressing a gas, and a second one of said units operates as a combustion and expansion chamber for converting energy released by combustion of a fuel in the gas into mechanical torque on a shaft (13), said transfer duct (5) serving to allow compressed gas being transferred from the compressor to the combustion and expansion chamber.

3. An internal combustion engine as claimed in claim 2, wherein the gas is a mixture of air and a fuel.

4. An internal combustion engine as claimed in claim 2, wherein the gas is air, and a fuel is added during the transfer of compressed gas from the compressor to the combustion and expansion chamber.

10. An internal combustion engine as claimed in any one of claims 1-4, wherein leakage of gasses past each said abutment (4, 10) is restricted by the geometry of the edge of the abutment (4, 10), and further by maintaining a narrow clearance between the abutment edge (22, 50, 51) and adjacent rotor (14, 26) and vane (3, 9) surfaces, the width of said narrow clearance being as small as possible without causing interference between the abutment edge (22, 50, 51) and adjacent rotor (14, 26) and vane (3, 9) surfaces during operation of the engine.

11. An internal combustion engine as claimed in any one of claims 1-4, wherein leakage of gasses past said vane (3, 9) is restricted by the geometry of the edge of the vane (3, 9), and further by maintaining a narrow clearance between the edge of the vane (3, 9) and an interior surface of a rotor housing, the width of said narrow clearance being as small as possible without causing interference between the edge of the vane (3, 9) and the interior surface of the rotor housing during operation of the engine.

12. An internal combustion engine as claimed in any one of claims 1-4, further comprising a system of one or more gears and shafts adapted to cause rotation of each said abutment in synchrony with respective ones of said rotors.

13. An internal combustion engine as claimed in claim 1, further comprising cooling means (34) for removing excessive heat from within said engine.

14. An internal combustion engine as claimed in claim 13, wherein said cooling means (34) comprises means for circulating a cooling fluid around at least a portion of the exterior of said engine.

15. An internal combustion engine as claimed in claim 13, wherein said cooling means (34) comprises means for circulating a cooling fluid around at least a portion of the interior of said engine.

16. An internal combustion engine as claimed in claim 13, 14 or 15, wherein said cooling fluid is air.

17. An internal combustion engine as claimed in claim 13, 14 or 15, wherein said cooling fluid is at least predominantly water.

18. An internal combustion engine as claimed in any one of claims 1-4, wherein combustion of a fuel within said engine in initiated by means of a spark-ignition system comprising a spark plug (11) operatively disposed within a portion of the transfer duct (5).

19. An internal combustion engine as claimed in claim 2, 3 or 4, further comprising a poppet-type valve mounted in operative relation to the opening (7) between the compressor and the transfer duct (5), said popet-type valve being adapted to control the flow of gas from the compressor and into the transfer duct.

## STATEMENT UNDER ARTICLE 19

Claim 1 has been amended to more clearly distinguish the presently claimed invention over the teachings of the prior art references cited in the International Search Report. Claims 10 and 11 have been amended to more clearly define features of the present invention. New claim 19 has been introduced to define further features of the present invention. The other claims remain unchanged.

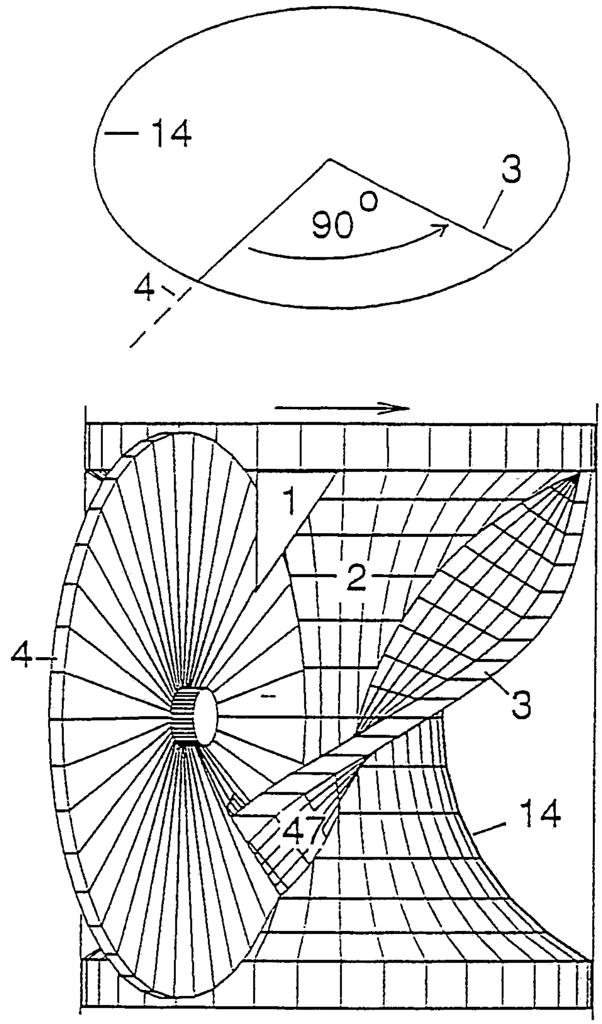


Figure 1

→ Direction of rotation

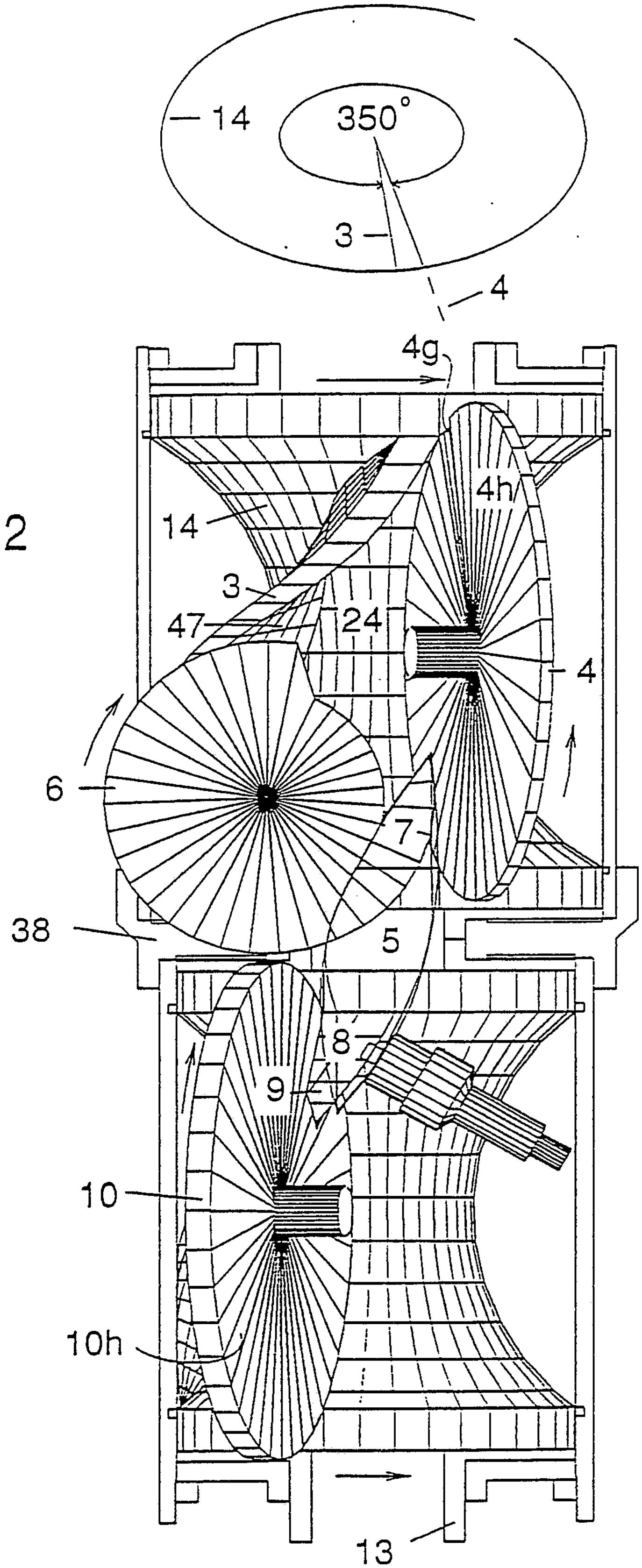


Figure 2

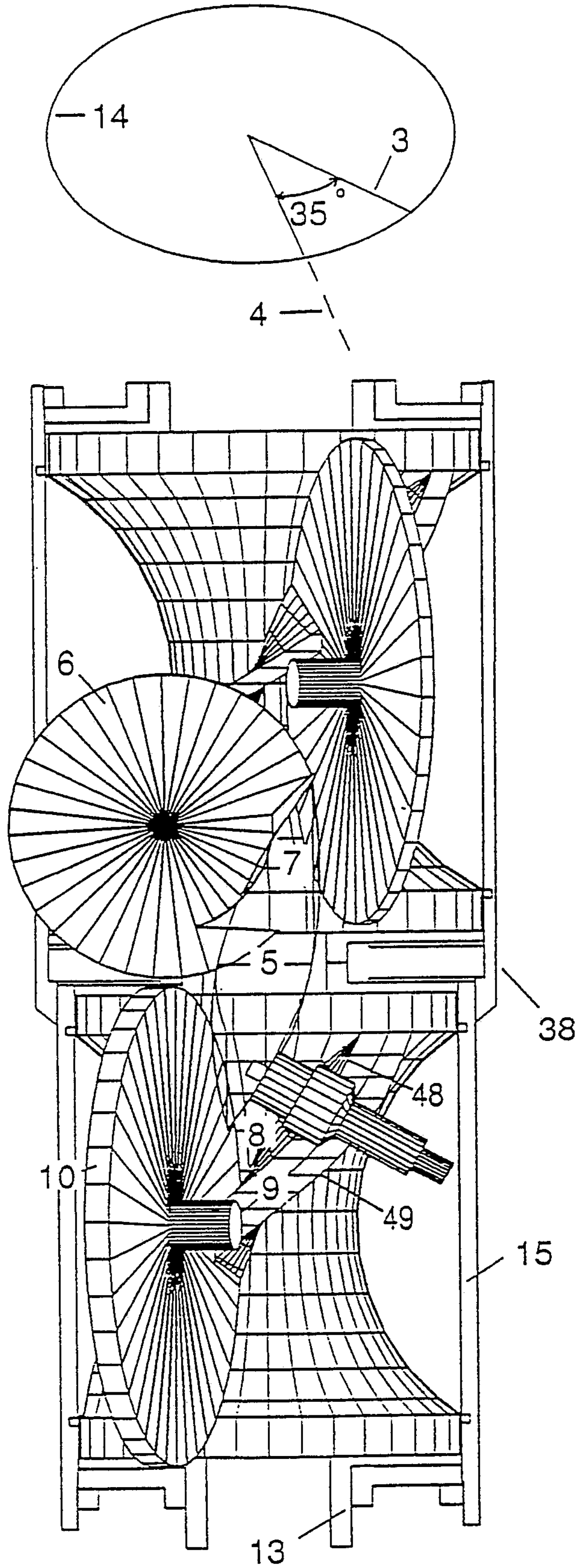


Figure 3

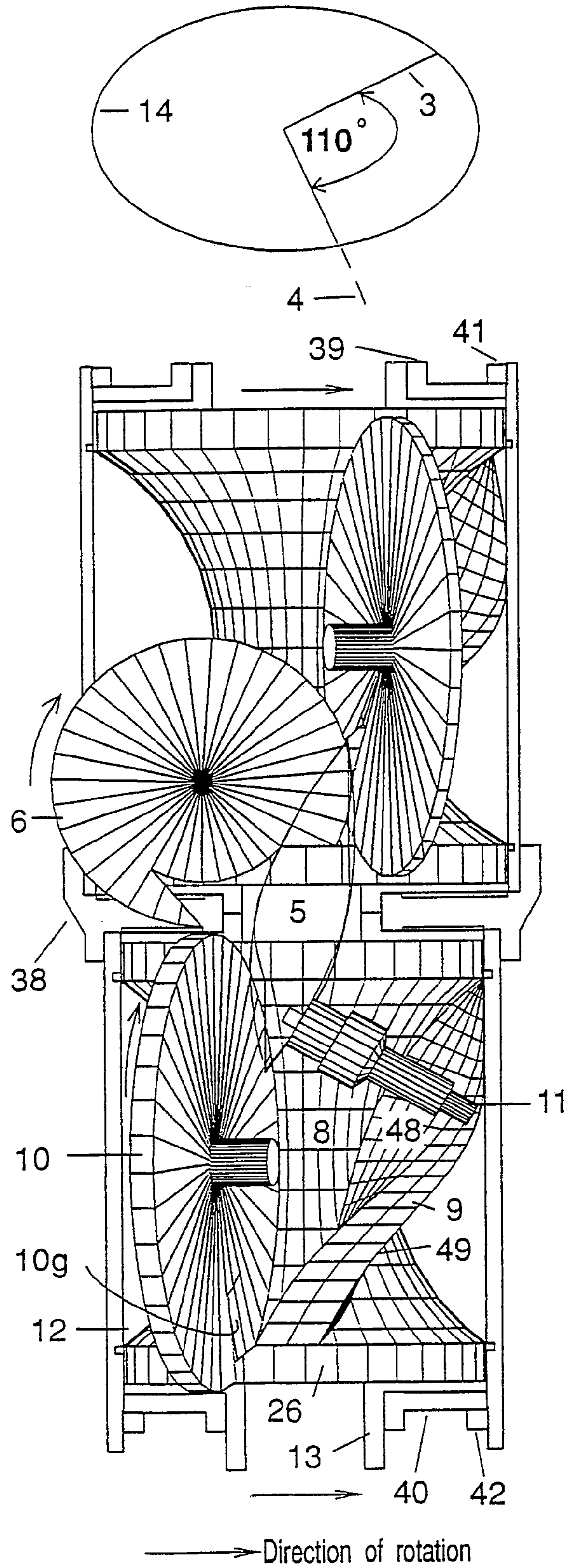
14
110°
3
4
39
41
6
5
38
11
8
48
10
9
49
10g
12
26
13
40
42
Direction of rotation

Figure 4

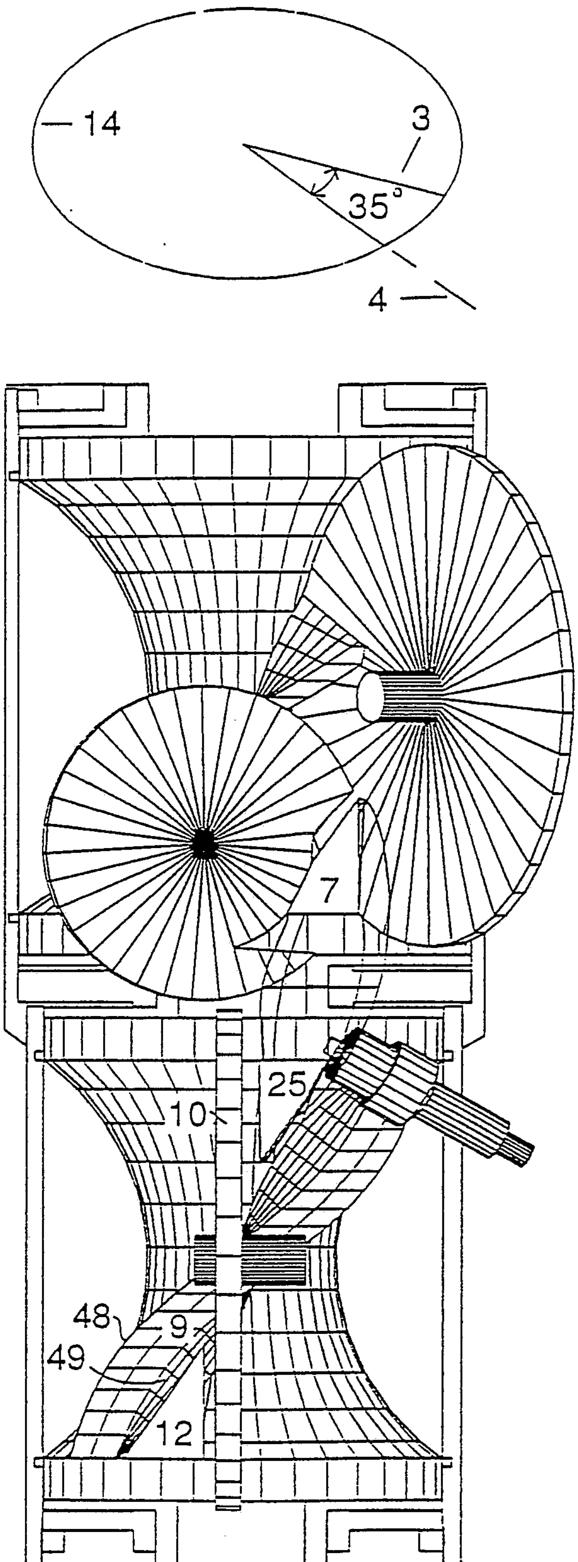


Figure 5

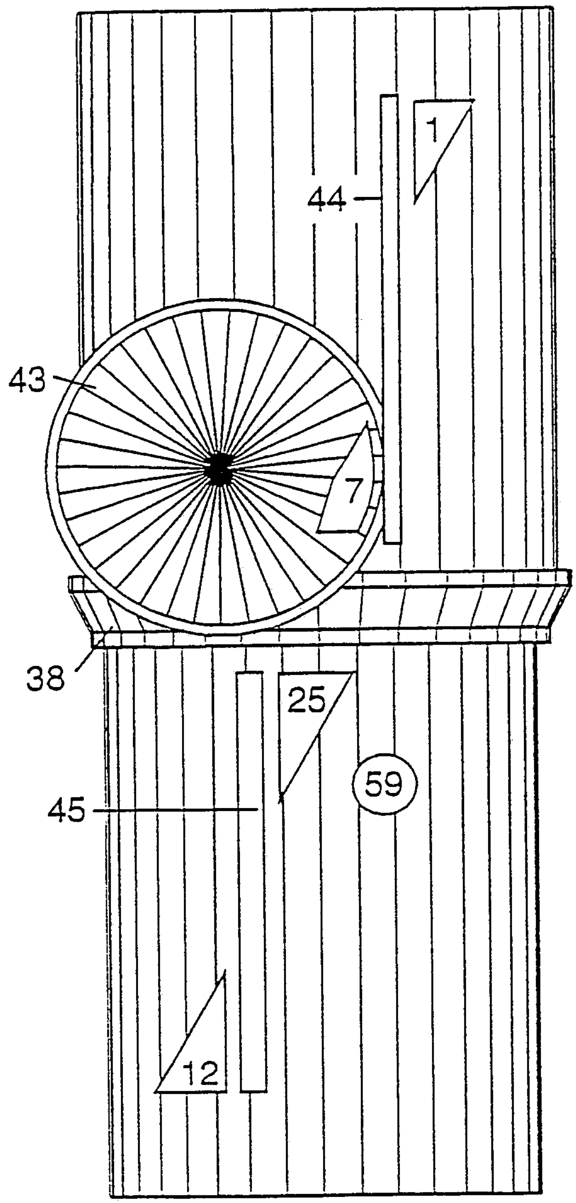


Figure 6

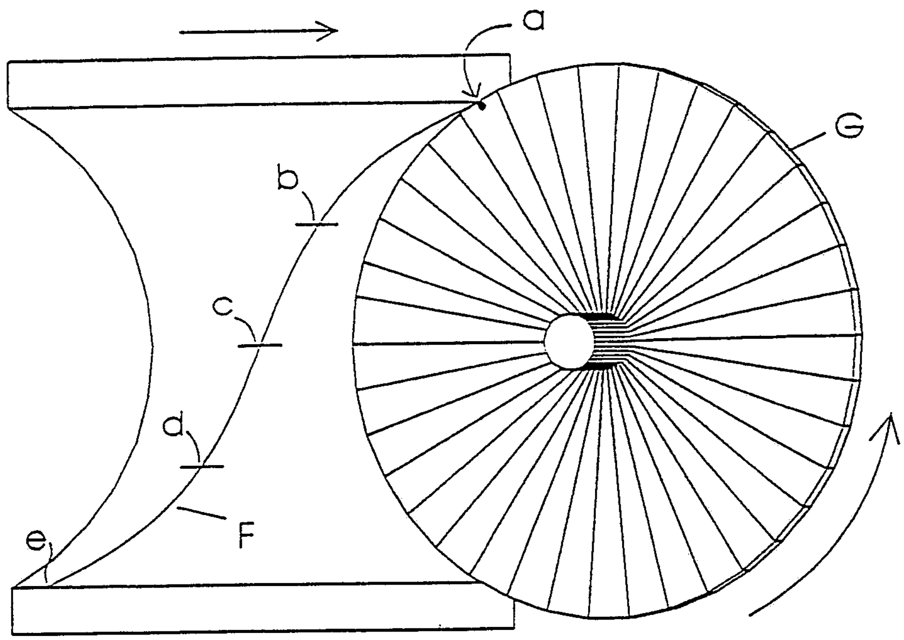


Figure 7

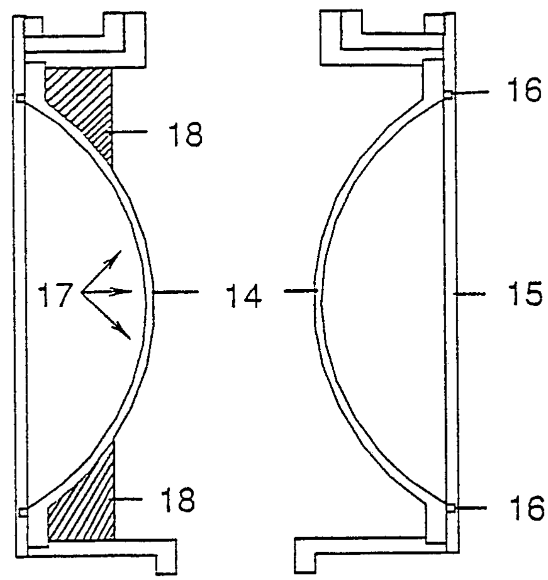


Figure 9

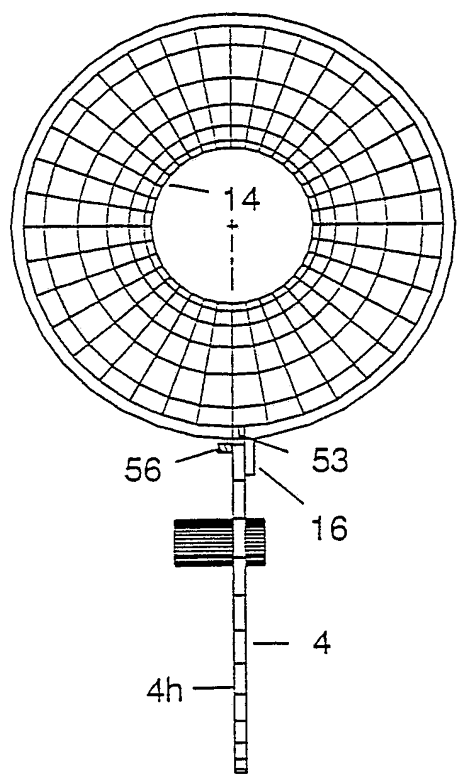


Figure 8

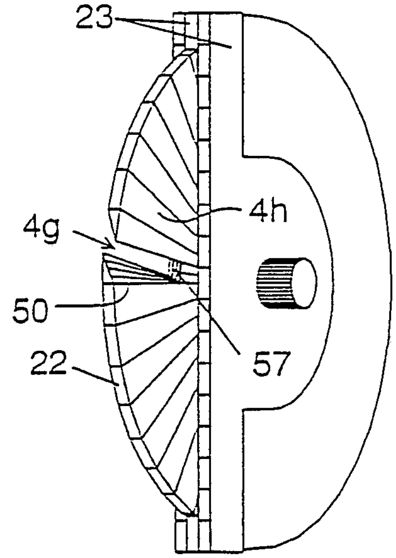


Figure 10

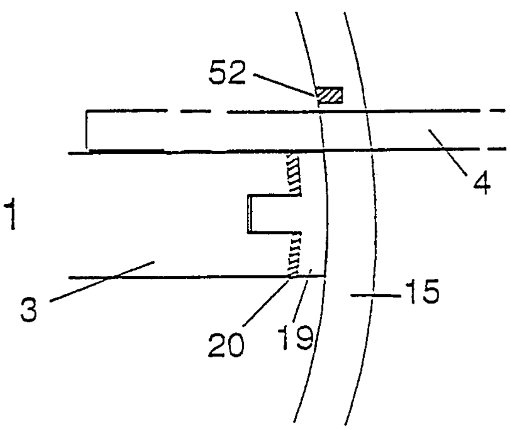


Figure 11

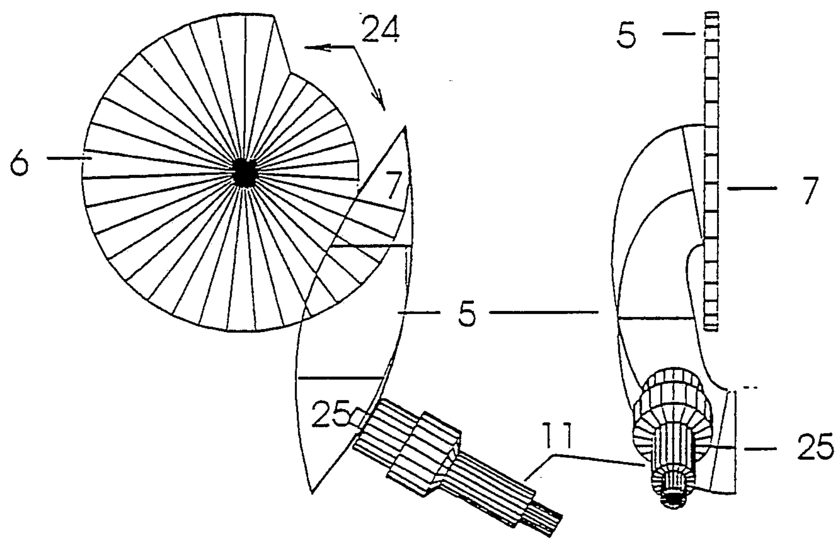


Figure 12 Figure 13

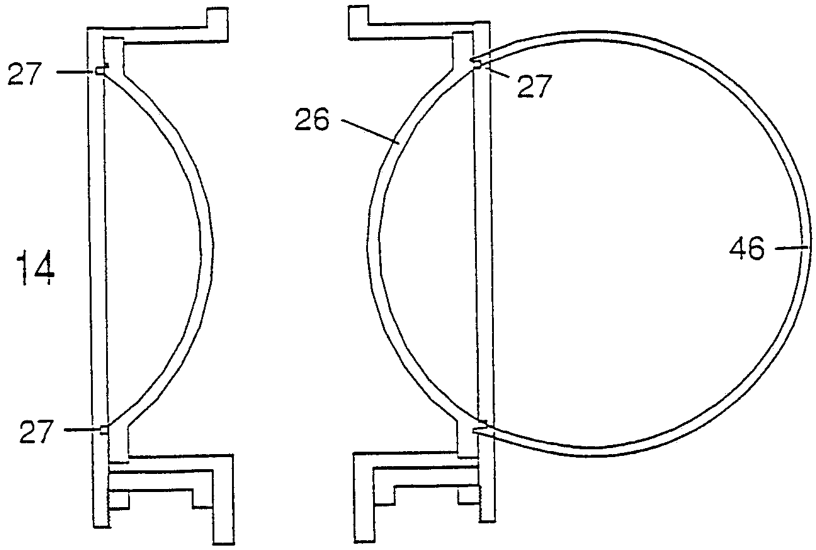


Figure 14

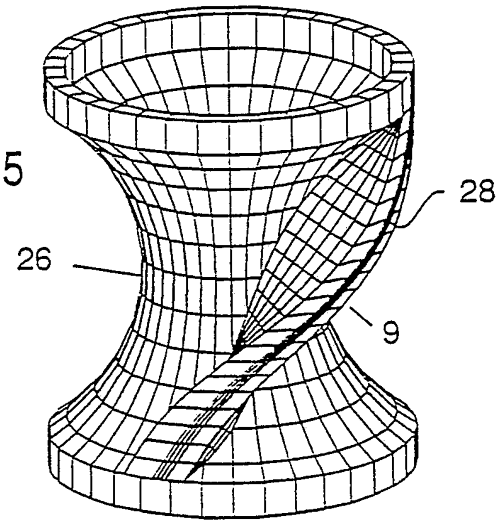


Figure 15

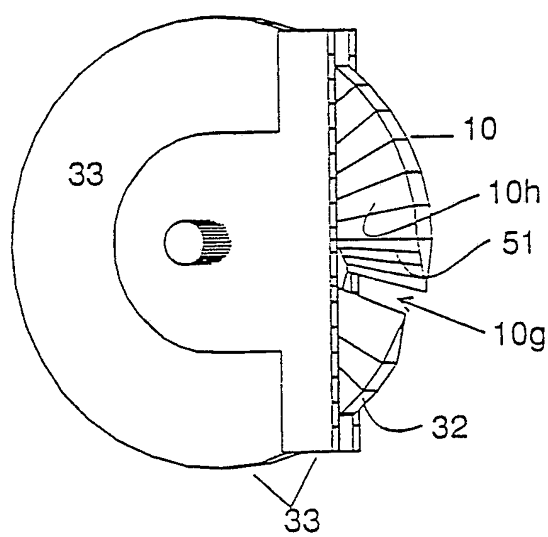


Figure 16

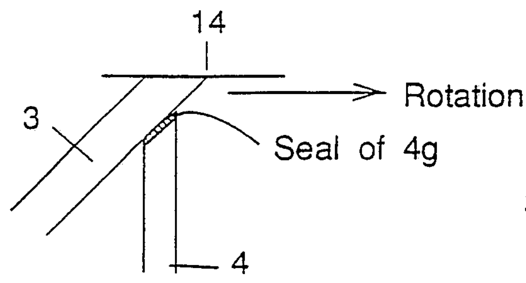


Figure 17a

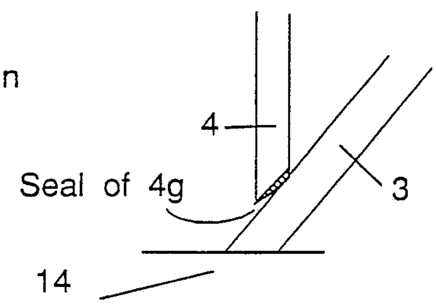


Figure 17b

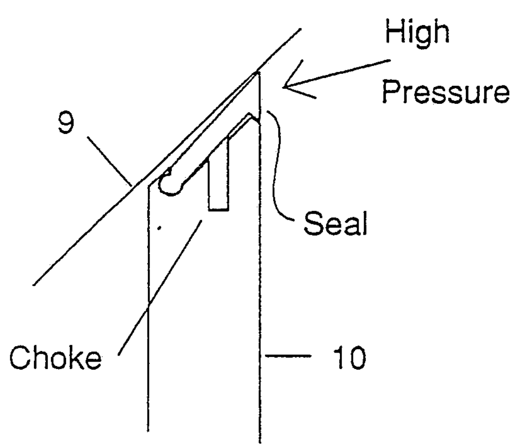


Figure 17c

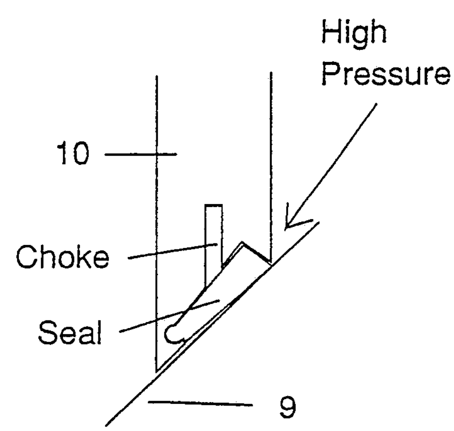


Figure 17d

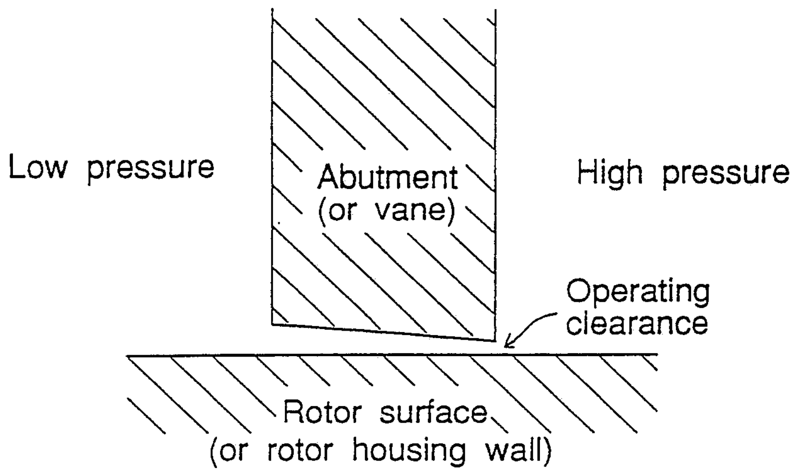


Figure 18a

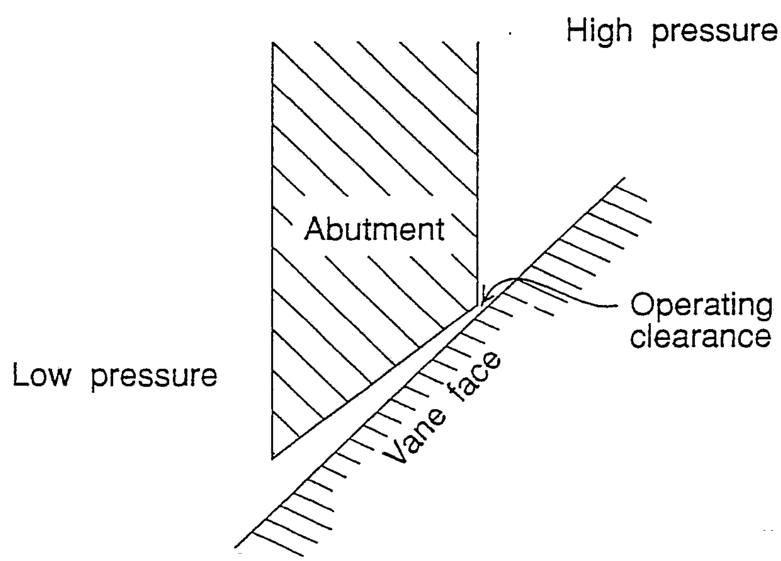


Figure 18b

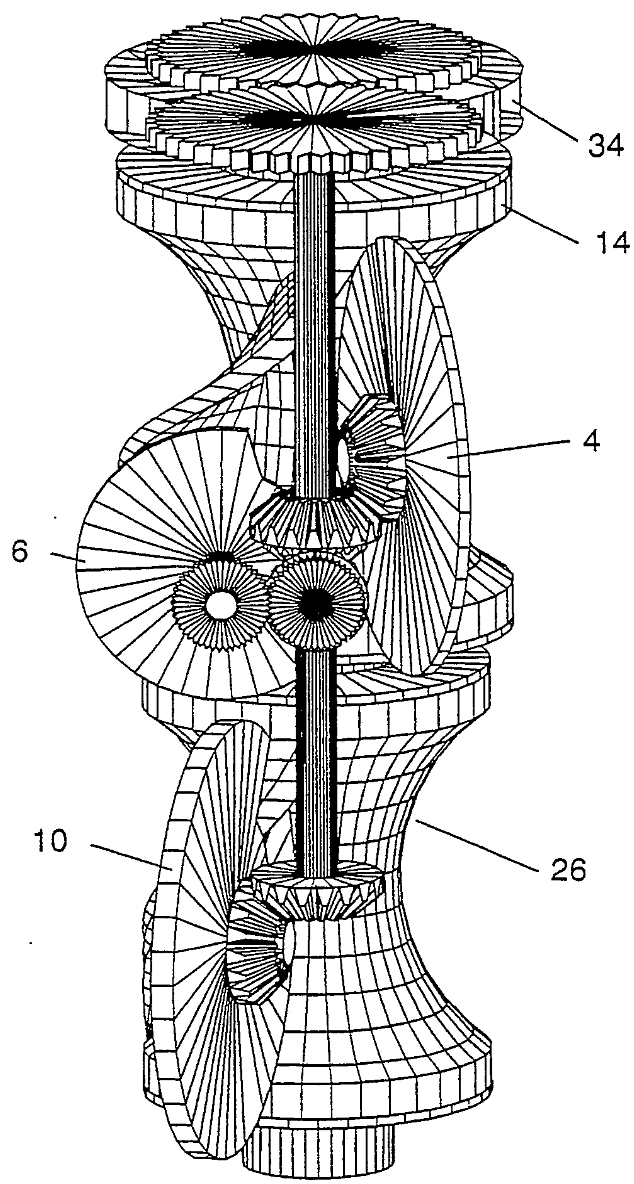


Figure19

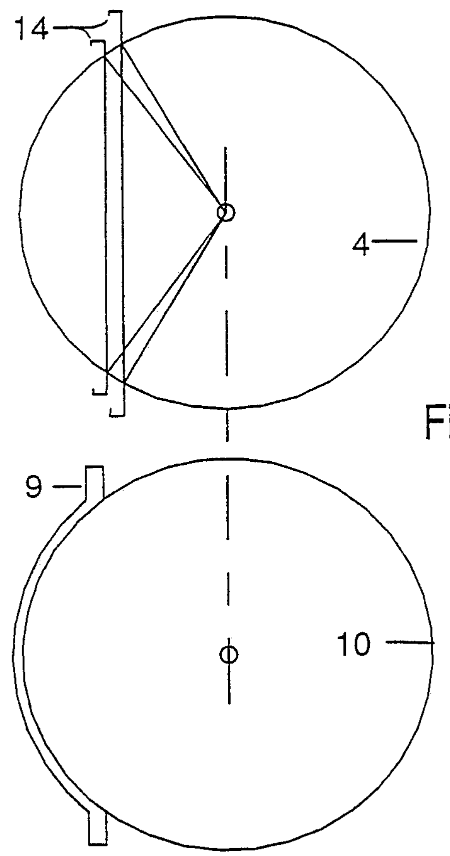


Figure 20

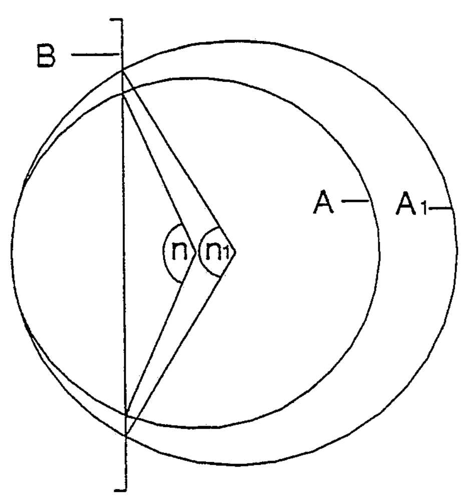


Figure 21

# INTERNATIONAL SEARCH REPORT

International Application No PCT/CA 93/00005

**I. CLASSIFICATION OF SUBJECT MATTER** (if several classification symbols apply, indicate all)[6]

According to International Patent Classification (IPC) or to both National Classification and IPC

Int.Cl. 5 F01C3/02

**II. FIELDS SEARCHED**

Minimum Documentation Searched[7]

| Classification System | Classification Symbols |
|---|---|
| Int.Cl. 5 | F01C ; F04C |

Documentation Searched other than Minimum Documentation to the Extent that such Documents are Included in the Fields Searched[8]

**III. DOCUMENTS CONSIDERED TO BE RELEVANT**[9]

| Category ° | Citation of Document, [11] with indication, where appropriate, of the relevant passages [12] | Relevant to Claim No.[13] |
|---|---|---|
| Y | FR,A,1 600 666 (JEANDEL) 27 July 1970 see the whole document | 1-4,6 |
| Y | FR,A,2 121 740 (NORTHWESTERN UNIVERSITY) 25 August 1972 see page 3, line 25 - page 13, line 21; figures | 1-4,6 |
| A | US,A,3 908 359 (JEANDEL) 30 September 1975 | |
| A | US,A,2 716 861 (GOODYEAR) 6 September 1955 | |

° Special categories of cited documents : [10]

"A" document defining the general state of the art which is not considered to be of particular relevance

"E" earlier document but published on or after the international filing date

"L" document which may throw doubts on priority claim(s) or which is cited to establish the publication date of another citation or other special reason (as specified)

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"X" document of particular relevance; the claimed invention cannot be considered novel or cannot be considered to involve an inventive step

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"&" document member of the same patent family

**IV. CERTIFICATION**

| Date of the Actual Completion of the International Search | Date of Mailing of this International Search Report |
|---|---|
| 16 MARCH 1993 | 24. 03. 93 |
| **International Searching Authority** | **Signature of Authorized Officer** |
| EUROPEAN PATENT OFFICE | MOUTON J.M.M.P. |

## ANNEX TO THE INTERNATIONAL SEARCH REPORT ON INTERNATIONAL PATENT APPLICATION NO.

CA 9300005
SA 68998

This annex lists the patent family members relating to the patent documents cited in the above-mentioned international search report.
The members are as contained in the European Patent Office EDP file on
The European Patent Office is in no way liable for these particulars which are merely given for the purpose of information. 16/03/93

| Patent document cited in search report | Publication date | Patent family member(s) | Publication date |
|---|---|---|---|
| FR-A-1600666 | 27-07-70 | None | |
| FR-A-2121740 | 25-08-72 | CA-A- 945966 | 23-04-74 |
| | | CH-A- 564146 | 15-07-75 |
| | | DE-A- 2201379 | 27-07-72 |
| | | GB-A- 1376908 | 11-12-74 |
| | | US-A- 3726616 | 10-04-73 |
| US-A-3908359 | 30-09-75 | FR-A- 2118279 | 28-07-72 |
| | | FR-A- 2162701 | 20-07-73 |
| | | DE-A,B,C 2162426 | 29-06-72 |
| | | GB-A- 1364638 | 21-08-74 |
| | | US-A- 3810722 | 14-05-74 |
| US-A-2716861 | | None | |

EPO FORM P0479

**PUB-NO:** WO009314299A1
**DOCUMENT-IDENTIFIER:** WO 9314299 A1
**TITLE:** ROTARY ENGINE
**PUBN-DATE:** July 22, 1993

**INVENTOR-INFORMATION:**

| NAME | COUNTRY |
|---|---|
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**ASSIGNEE-INFORMATION:**

| NAME | COUNTRY |
|---|---|
| BELANGER J ROBERT | CA |

**APPL-NO:** CA09300005
**APPL-DATE:** January 12, 1993

**PRIORITY-DATA:** CA02059757A (January 21, 1992)

**INT-CL (IPC):** F01C003/02

**EUR-CL (EPC):** F01C003/02

**US-CL-CURRENT:** 418/195 , 418/200

**ABSTRACT:**

An internal combustion engine composed of separate compression and combustion/expansion chambers (8) linked by a transfer duct (5), is disclosed. The compressor and expansion chambers include

similarly shaped rotors (14, 26) and abutments (4, 10). Each of the rotors (14, 26) includes a vane (3, 9), and the rotors (14, 26) and abutments (4, 10) are rotationally coupled such that the vane (3, 9) enters into operative engagement with the associated abutment (4, 10) once during each revolution of the rotor (14, 26). Each abutment (4, 10) consists of a rotating disk having a gap (4g, 10g) which permits passage of the vane (3, 9) through the abutment (4, 10).